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(11) **EP 0 974 016 B1**

(12) **EUROPEAN PATENT SPECIFICATION**

(45) Date of publication and mention
of the grant of the patent:
27.03.2002 Bulletin 2002/13

(21) Application number: **98915421.6**

(22) Date of filing: **09.04.1998**

(51) Int Cl.7: **F16H 55/08**

(86) International application number:
PCT/US98/07112

(87) International publication number:
WO 98/45623 (15.10.1998 Gazette 1998/41)

(54) **GEAR FORM CONSTRUCTIONS**
AUSLEGUNG VON ZAHNRADPROFILEN
CONSTRUCTIONS DE FORMES D'ENGRENAGE

(84) Designated Contracting States:
AT BE CH DE DK ES FR GB IT LI NL SE

(30) Priority: **10.04.1997 US 43165 P**
08.04.1998 US 57170

(43) Date of publication of application:
26.01.2000 Bulletin 2000/04

(73) Proprietor: **Genesis Partners, L.P.**
Horsham, PA 19044 (US)

(72) Inventors:
• **BERLINGER, Bernard, E., Jr.**
Furlong, PA 18925 (US)

• **COLBOURNE, John, Robert**
St. Albert, Alberta T8N 4L6 (CA)

(74) Representative: **Pacitti, Paolo et al**
Murgitroyd and Company
165-169 Scotland Street
Glasgow G5 8PL (GB)

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US-A- 3 937 098 **US-A- 5 271 289**

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EP 0 974 016 B1

Description**FIELD OF THE INVENTION**

5 **[0001]** The present invention relates to the construction of profiles of conjugate gears. In particular, the present invention may be applied directly to the design and construction of spur gears, or as the transverse section profiles of helical gears.

BACKGROUND OF THE INVENTION

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[0002] Several attempts have been made to reduce the contact stress in gear teeth, since a high contact stress causes pitting and eventually failure of the teeth. Many of these attempts have been based on the Wildhaber or Novikov types of gearing, in addition to the more common involute forms. In Novikov gears, the profiles in the transverse sections are circular arcs, the pinion being convex and the gear being concave. The problem with circular arc profiles is that they are not conjugate. With conjugate profiles, the angular velocity ratio is exactly constant. In each transverse section there is continuous contact throughout the meshing cycle, and the contact point moves along each profile, towards the tip of the driving tooth and towards the root of the driven tooth. U.S. 3,937,098 to Rouverol provides an example of gearing profiles that are not conjugate according to the preamble of claim 1.

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[0003] Since a variable angular velocity ratio is not acceptable, Novikov gears are always made helical. This means that in each transverse section, only one pair of points ever come into contact. At any instant, there will be one transverse section where contact occurs, and this contact point moves axially along the tooth face as the gears rotate. Such gears are generally noisy, and have not been widely used. The same comments apply to Wildhaber gears, which are shaped as circular arcs in the normal sections. More recently, other profiles have been proposed, in which there are two or more contact points in each transverse section. They still suffer, though, from the disadvantages just described.

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SUMMARY OF THE INVENTION

[0004] A gearing system, constructed in accordance with the present invention, includes a first gear having a first plurality of teeth each having a first tooth profile and a mating gear having a second plurality of teeth each having a second tooth profile. The first tooth profile of the first plurality of teeth of the first gear includes a first transition zone disposed between a first concave portion lying within the dedendum of the first gear and a first convex portion lying within the addendum of the first gear. The second tooth profile of the second plurality of teeth of the second gear includes a second transition zone disposed between a second concave portion lying within the dedendum of the mating gear and a second convex portion lying within the addendum of the mating gear. The second convex portion of second tooth profile of the second plurality of teeth of the second gear is conjugate to the first concave portion of the first tooth profile of the first plurality of teeth of the first gear and the second concave portion of the second tooth profile of the second plurality of teeth of the second gear is conjugate to the first convex portion of the first tooth profile of the first plurality of teeth of the first gear.

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BRIEF DESCRIPTION OF THE DRAWINGS**[0005]**

Figure 1 shows conjugate profiles of a first gearing system constructed in accordance with the present invention.

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Figure 2 shows conjugate profiles of a second gearing system constructed in accordance with the present invention.

Figures 3A through 3D show a variety of types of gearing systems in which the present invention may be incorporated.

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DETAILED DESCRIPTION OF THE INVENTION

[0006] Referring to Figure 1, a gearing system, constructed in accordance with the present invention, includes a first gear **10** having a first plurality of teeth each having a first tooth profile and a mating gear **12** having a second plurality of teeth each having a second tooth profile. The first profile of the first plurality of teeth of the first gear **10** includes a first transition zone **10a** disposed between a first concave portion **10b** lying within the dedendum of the first gear and a first convex portion **10c** lying within the addendum of the first gear. The second profile of the second plurality of teeth of mating gear **12** includes a second transition zone **12a** disposed between a second concave portion **12b** lying within

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the dedendum of the mating gear and a second convex 12c portion lying within the addendum of the mating gear.

[0007] The second concave portion 12b of the second profile of the second plurality of teeth of mating gear 12 is conjugate to the first convex portion 10c of the first tooth profile of the first plurality of teeth of first gear 10 and the second convex portion 12c of the second profile of the second plurality of teeth of mating gear 12 is conjugate to the first concave portion 10b of the first tooth profile of the first plurality of teeth of the first gear. In addition, the second transition zone 12a of the second profile of the second plurality of teeth of mating gear 12 can be conjugate to the first transition zone 10a of the first tooth profile of the first plurality of teeth of first gear 10, so that the second tooth profile of the mating gear is conjugate to the first tooth profile of the first gear. This means that when the gears are in mesh, the angular velocity ratio is constant.

[0008] A pair of conjugate profiles can be defined by either the pinion tooth profile, the gear tooth profile, the basic rack profile, or the shape of the contact path. When any one of these four shapes is known, it is possible to calculate the other three shapes. The methods by which these shapes are found have been described by Buckingham, *Analytical Mechanics of Gears*, McGraw-Hill, New York, 1949, republished by Dover, New York, 1963, incorporated in its entirety herein. The most common method for defining a pair of profiles is to choose the shape of the basic rack. For example, if the basic rack profile is straight, involute gears are obtained. Less commonly, the shape of one tooth profile is chosen. For example, in Gerotor internal gear pumps, the lobes of the outer rotor are circular. Less commonly still, the shape of the contact path is chosen. For example, cycloidal gears can be defined as having profiles for which the contact path consists of two circular arcs.

[0009] Consider the following method for designing conjugate gear tooth profiles. There are two important properties of conjugate profiles, which are described by Buckingham. First, they must satisfy the Law of Gearing, which states that the common normal at the contact point (also known as the line of action) always passes through the pitch point. Secondly, the radii of curvature ρ_1 and ρ_2 of the profiles must satisfy the Euler-Savary equation,

$$\frac{1}{\rho_1 - S} + \frac{1}{\rho_2 + S} = \frac{1}{\sin \phi} \left(\frac{1}{R_{p1}} + \frac{1}{R_{p2}} \right), \quad (1)$$

where R_{p1} and R_{p2} are the pitch circle radii, ϕ is the gear pair pressure angle, namely the angle between the line of action and the line through the pitch point perpendicular to the line of centers; S is the distance from the pitch point to the contact point, positive when the contact point lies on one side of the line of centers, and negative on the other; ρ_1 and ρ_2 are the radii of curvature, positive for convex profiles and negative for concave profiles. The reciprocals of the radii of curvature, $1/\rho_1$ and $1/\rho_2$, are known as the curvatures.

[0010] It is important to note that whenever contact between meshing teeth is referred to herein it is understood that in fact for practical reasons meshing teeth of gears often only make contact through a thin lubricating film. Therefore, distinctions between 'contact' and 'no contact' are made based on a standard of thickness of and transmission of forces through such thin lubricating films.

[0011] If the relative curvature is to remain constant, the radii of curvature should satisfy the following relation,

$$\frac{1}{\rho_1} + \frac{1}{\rho_2} = \text{constant} . \quad (2)$$

[0012] In other words, the gearing system shown in, for example, Figure 1, or the like, may be designed in accordance with the present invention and such that the relative curvature of the first tooth profile and the second tooth profile is a constant.

[0013] If, instead, the contact stress is to remain constant, the radii of curvature should satisfy the following relation,

$$\frac{1}{\cos \phi} \left(\frac{1}{\rho_1} + \frac{1}{\rho_2} \right) = \text{constant} . \quad (2a)$$

[0014] In other words, the gearing system shown, for example, in Figure 1, or the like, may be designed in accordance

with the present invention and such that the contact stress between meshing teeth of the first gear 10 with the mating gear 12 is a constant.

[0015] Equations (2) and (2a) may be replaced by any predetermined mathematical function set to a constant and parameterized (as are equation (2) and (2a)) by the radius of curvature associated with the first tooth profile and the radius of curvature associated with the second tooth profile. Equations (1) and (2), (1) and (2a), or (1) and any predetermined mathematical function (parameterized by the radii of curvature) set to a constant, may be solved to find the values of p_1 and p_2 , whenever the values of S and ϕ are known.

[0016] The profiles of the two gears can now be found in the following manner. Choose an initial point $S = S_0$, $\phi = \phi_0$ on the contact path near the pitch point, where S and ϕ are the polar coordinates of the point. The radii R_1 and R_2 of the two profile points, and the corresponding profile angles ϕ_1 and ϕ_2 , can now be found by conventional methods.

[0017] Once S and ϕ are known, the values of p_1 and p_2 can be calculated. It is now possible to construct a small increment of the tooth profile of the first gear as a circular arc of radius p_1 . Then the radius R_1 , and the profile angle ϕ_1 at the end of the increment are calculated, and using the conventional theory of conjugate profiles the corresponding values of R_2 , ϕ_2 , S and ϕ may also be calculated. The process is repeated as often as desired to construct the addendum of the first gear and the dedendum of the mating gear. Since the coordinates of a number of points on the contact path have been found, the shape of the basic cutter can be deduced by conventional methods. A family of gears is composed of gears that are conjugate to a basic cutter. The basic rack is the complement of the basic cutter, and the tooth profile of the basic rack is therefore the same as that of a gear belonging to the family with an infinite number of teeth.

[0018] To find the shape of the dedendum of the first gear and the addendum of the mating gear, the entire procedure explained above is repeated, starting from an initial point on the contact path on the opposite side of the pitch point. A suitable initial point is given by $S = -S_0$, $\phi = \phi_0$, but it is not essential to use these values. The recommended value of S_0 lies between 0.1 modules and 0.5 modules, where the module is a length defined in terms of the center distance C and the tooth numbers N_1 and N_2 , as $2C/(N_1 + N_2)$, while ϕ_0 can be any value such that the profile angles ϕ_1 and ϕ_2 are positive at both initial points.

[0019] The above procedure describes a design for the addendum and the dedendum of each gear, but there is a short section of each profile between the addendum and the dedendum the shape of which is not yet determined. This section is called the transition zone.

[0020] The transition zone profiles can be designed in one of three ways. If the two initial points and the pitch point are collinear, the transition zone section of the basic rack corresponding to a gear can be formed by a straight line, so that the gear tooth profiles are involute in their transition zones. If the two initial points and the pitch point are not collinear, the dedendum and addendum sections of the basic rack can be joined by any smooth curve, such as a cubic curve, which provides continuity of profile and profile slope at the ends of the transition zones. Hence, the transition zone of a gear is formed by a smooth curve in the profile of the basic rack that corresponds to the gear.

[0021] If the gear tooth transition zones are designed by either of the methods just described, then the two gear tooth profiles will be conjugate along their entire lengths. The profiles can therefore be used for spur gears, for example, or the like, or as the transverse section profiles of helical gears, for example, or the like. These profiles suffer, however, from the disadvantage that the relative curvature, and therefore also the contact stress, is generally much higher in the transition zones than in the addendum or dedendum.

[0022] Figure 1 shows a 21-tooth pinion and a 72-tooth gear, with involute transition zones 10a and 12a. In the addendum and dedendum, the contact stress is equal to 0.75 times the reference value, where the reference value is the contact stress at the pitch point of a 20° pressure angle of an involute gear pair, with the same tooth numbers, center distance, face width and input torque. At the pitch circles, however, in the middle of the transition zones, the contact stress is equal to 1.76 times the reference value. Thus the advantage of the constant contact stress design is lost.

[0023] The third design method, which is the preferred option, is to design the profiles so that there is no contact within the transition zones. In this case contact between meshing teeth is made only along the convex addendum and concave dedendum portions. For example, a gear and pinion system may be designed so that contact is made between the convex addendum portion of the gear and the concave dedendum portion of the pinion and between the convex addendum portion of the pinion and the concave dedendum portion of the gear. The profiles are therefore not conjugate in the transition zones. Hence, in order to maintain a constant ratio of the angular velocity of the first gear to the angular velocity of the mating gear, the gears should be helical with transverse section profiles designed in accordance with the present invention.

[0024] Figure 2 shows the transverse section profiles of a 15-tooth pinion and a 90-tooth gear, which may be designed so that there no contact is made along the transition zones 20a and 22a. It is important to note that due to the scale of Figure 2, and the slight and subtle nature of the transition zone it is impossible to discern simply by observing Figure 2 the fact that no contact is made, at the flanks which are in contact, along the transition zone. The addendum 20c and 22c, and dedendum 20b and 22b portions of these profiles are designed using Equations (1) and (2a), with the maximum relative curvature equal to 51% of the reference value.

[0025] Since there is not continuous contact along the profile from the form circle to the tip circle, it is not possible to define the contact ratio in the conventional manner. A contact ratio for the dedendum, and a contact ratio for the addendum may be calculated, and the sum of these quantities will be called the total transverse contact ratio. The value of the total transverse contact ratio depends on the tooth numbers and the required contact stress. Table 1 gives the lowest achievable relative curvature, as a percentage of the reference value, if the total transverse contact ratio is to be at least 1.2.

Table 1

Lowest Achievable Relative Curvature, as a Percentage of the Reference Value, giving a Total Transverse Contact Ratio of at least 1.2.				
N ₂	40	60	80	100
N ₁				
15	52	51	51	51
20	51	51	49	49
30	51	49	49	49
40	49	49	49	49

[0026] If the transverse profiles of a gear pair are designed according to the procedure described above, using Equations (1) and (2a), then the contact stress will be constant provided the gears are spur gears. The situation in a helical gear pair is more complicated, since the contact stress depends on both the relative curvature in the normal section, and on the load intensity. The load intensity depends in turn on the lengths of the contact lines, which vary throughout the meshing cycle. The length of the contact lines may also be reduced when there is no contact in the transition zone. For profiles of this type there is no simple formula for calculating the contact stress. The contact stress, however, may be calculated numerically. The following conclusions may be drawn from such a calculation. When the transverse profiles are designed in accordance with the present invention, using either Equation (2) or Equation (2a), the relative curvature may be kept to about half the reference value, while still maintaining an adequate contact ratio (see Table 1). The contact stress is then approximately constant at all points of contact throughout the meshing cycle, and the maximum value is significantly less than the maximum contact stress that would occur in an involute gear pair with the same overall dimensions.

[0027] The method of profile construction described above may also be used to design the tooth profiles of an internal gear pair.

[0028] Alternatively, an internal gear may also be designed by conventional methods with a tooth profile which is conjugate to an external gear designed by the method described above. This procedure may be required in order to design a planetary gearset, in which the sun and the planet would be designed by the method described above, and the internal gear would be designed with a tooth profile conjugate to that of the planet.

[0029] It should be noted that gears constructed in accordance with the present invention may be cut by conventional methods. The external gears can be hobbled, generating ground or form ground, and the internal gears can be shaped, using a pinion cutter.

[0030] A basic rack designed in accordance with the present invention can be used, for example, as the profile of a crown gear, which defines the tooth profiles of a pair of bevel gears.

[0031] The present invention can be incorporated in various different gearing systems. In a the spur gearing system shown in Figure 3A, the axis of rotation of first gear 10 is parallel to the axis of rotation of mating gear 12. In the crossed helical gearing system of Figure 3B, the axis of rotation of first gear 40 is perpendicular to the axis of rotation of mating gear 42. In the hypoid gearing system of Figure 3C, the axis of rotation of first gear 50 does not intersect the axis of rotation of mating gear 52. In the spiral bevel gearing system of Figure 3D, the axis of rotation of first gear 60 is at an angle to the axis of rotation of mating gear 62.

Claims

1. A gearing system comprising:

a first gear (10) having a first plurality of teeth each having a first tooth profile including a first transition zone (10a) disposed between a first concave portion (10b) lying within the dedendum of said first gear (10) and a first convex portion (10c) lying within the addendum of said first gear (10); and

a mating gear (12) having a second plurality of teeth each having a second tooth profile including a second transition zone (12a) disposed between a second concave portion (12b) lying within the dedendum of said mating gear (12), and a second convex portion (12c) lying within the addendum of said mating gear (12),
characterized by:

(1) said first tooth profile of said first plurality of teeth of said first gear (10) being conjugate to said second tooth profile of said second plurality of teeth of said mating gear (12); except in said first and said second transition zones (10a, 12a), with said second concave portion (12b) conjugate to said first convex portion (10c) and said second convex portion (12c) conjugate to said first concave portion (10b), and

(2) the relative curvature of said first tooth profile of said first plurality of teeth of said first gear (10) and said second tooth profile of said second plurality of teeth of said mating gear (12) being a constant which is less than the relative curvature of 20° pressure angle involute gears at their pitch point.

2. The gearing system of claim 1, **characterized by** a predetermined mathematical function parameterized by a radius of curvature associated with said first tooth profile of said first plurality of teeth of said first gear (10) and a radius of curvature associated with said second tooth profile of said second plurality of teeth of said mating gear (12) being a constant.

3. The gearing system of claim 1, **characterized by** the maximum contact stress between meshing of said first plurality of teeth of said first gear (10) with said second plurality of teeth of said mating gear (12) being a constant.

4. The gearing system of claim 1, **characterized by** the axis of rotation of said first gear (10) being parallel to the axis of rotation of said mating gear (12).

5. The gearing system of claim 1, **characterized by** the axis of rotation of said first gear (10) being at an angle to the axis of rotation of said mating gear (12).

6. The gearing system of claim 1, **characterized by** said first tooth profile of said first plurality of teeth of said first gear (10) being a first transverse section profile and said second tooth profile of said second plurality of teeth of said mating gear (12) being a second transverse section profile.

7. The gearing system of claim 1, **characterized by** contact between meshing of said first plurality of teeth of said first gear (10) with said second plurality of teeth of said mating gear (12) being made along:

(a) said first concave portion (10b) of said first tooth profile of said first plurality of teeth of said first gear (10) and said second convex portion (12c) of said second tooth profile of said second plurality of teeth of said mating gear (12), and

(b) said first convex portion (10c) of said first tooth profile of said first plurality of teeth of said first gear (10) and said second generally concave portion (12b) of said second tooth profile of said second plurality of teeth of said mating gear (12).

8. The gearing system of claim 7, **characterized by** said first gear (10) and said mating gear (12) being helical.

9. The gearing system of claim 8, **characterized by** a ratio of the angular velocity of said first gear (10) to the angular velocity of said mating gear (12) being a constant.

10. The gearing system of claim 1, **characterized by** said first gear (10) having a basic rack profile having a transition zone corresponding to said first transition zone (10a) of said first tooth profile of said first plurality of teeth of said first gear (10) which is a smooth curve.

11. The gearing system of claim 10, **characterized by** said smooth curve of said transition zone of said basic rack profile being a straight line.

12. The basic rack profile of claim 10, **characterized by** said smooth curve of said transition zone of said basic rack profile being a cubic curve.

13. The gearing system of claim 1, **characterized by** no contact between meshing of said first plurality of teeth of said first gear (10) with said second plurality of teeth of said mating gear (12) being made along said first transition zone (10a) and said second transition zone (12a).

14. The gearing system of claim 7, **characterized by** no contact between meshing of said first plurality of teeth of said first gear (10) with said second plurality of teeth of said mating gear (12) being made along said first transition zone (10a) and said second transition zone (12a).

Patentansprüche

1. Ein Getriebesystem, bestehend aus:

einem ersten Getriebe (10) mit einer ersten Vielzahl von Zähnen, wobei jeder Zahn ein erstes Zahnprofil aufweist, das einen ersten Übergangsbereich (10a) umfaßt, der zwischen einem ersten hohlen Abschnitt (10b), der sich innerhalb der Fußhöhe des ersten Getriebes (10) befindet, und einem ersten konvexen Abschnitt (10c), der sich innerhalb der Kopfhöhe des ersten Getriebes (10) befindet, angeordnet ist; und

einem Gegenrad (12) mit einer zweiten Vielzahl von Zähnen, wobei jeder Zahn ein zweites Zahnprofil aufweist, das einen zweiten Übergangsbereich (12a) umfaßt, der zwischen einem zweiten hohlen Abschnitt (12b), der sich innerhalb der Fußhöhe des Gegenrads (12) befindet, und einem zweiten konvexen Abschnitt (12c), der sich innerhalb der Kopfhöhe des Gegenrads (12) befindet, angeordnet ist, **dadurch gekennzeichnet, daß:**

(1) das erste Zahnprofil der ersten Vielzahl von Zähnen des ersten Getriebes (10) mit dem zweiten Zahnprofil der zweiten Vielzahl von Zähnen des Gegenrads (12) verbunden ist; außer in dem ersten und dem zweiten Übergangsbereich (10a, 12a), wobei der zweite hohle Abschnitt (12b) mit dem ersten konvexen Abschnitt (10c) verbunden ist und der zweite konvexe Abschnitt (12c) mit dem ersten hohlen Abschnitt (10b) verbunden ist, und

(2) die Relativkrümmung des ersten Zahnprofils der ersten Vielzahl von Zähnen des ersten Getriebes (10) und des zweiten Zahnprofils der zweiten Vielzahl von Zähnen des Gegenrads (12) ein Festwert ist, der weniger als die Relativkrümmung von Evolventenrädern mit einem Eingriffswinkel von 20° bei ihrem Wälzpunkt ist.

2. Getriebesystem gemäß Anspruch 1, **dadurch gekennzeichnet, daß** eine vorgegebene mathematische Funktion, die durch einen mit dem ersten Zahnprofil der ersten Vielzahl von Zähnen des ersten Getriebes (10) assoziierten Krümmungsradius und einen mit dem zweiten Zahnprofil der zweiten Vielzahl von Zähnen des Gegenrads (12) assoziierten Krümmungsradius parametrisiert ist, ein Festwert ist.

3. Getriebesystem gemäß Anspruch 1, **dadurch gekennzeichnet, daß** die maximale Kontaktbelastung beim Eingreifen der ersten Vielzahl von Zähnen des ersten Getriebes (10) in die zweite Vielzahl von Zähnen des Gegenrads (12) ein Festwert ist.

4. Getriebesystem gemäß Anspruch 1, **dadurch gekennzeichnet, daß** die Drehachse des ersten Getriebes (10) parallel zur Drehachse des Gegenrads (12) ist.

5. Getriebesystem gemäß Anspruch 1, **dadurch gekennzeichnet, daß** die Drehachse des ersten Getriebes (10) in einem Winkel zur Drehachse des Gegenrads (12) liegt.

6. Getriebesystem gemäß Anspruch 1, **dadurch gekennzeichnet, daß** das erste Zahnprofil der ersten Vielzahl von Zähnen des ersten Getriebes (10) ein erstes Querschnittsprofil ist und das zweite Zahnprofil der zweiten Vielzahl von Zähnen des Gegenrads (12) ein zweites Querschnittsprofil ist.

7. Getriebesystem gemäß Anspruch 1, **dadurch gekennzeichnet, daß** der Kontakt beim Eingreifen der ersten Vielzahl von Zähnen des ersten Getriebes (10) in die zweite Vielzahl von Zähnen des Gegenrads (12) entlang folgendem erfolgt:

(a) dem ersten hohlen Abschnitt (10b) des ersten Zahnprofils der ersten Vielzahl von Zähnen des ersten

Getriebes (10) und dem zweiten konvexen Abschnitt (12c) des zweiten Profils der zweiten Vielzahl von Zähnen des Gegenrads (12), und

(b) dem ersten konvexen Abschnitt (10c) des ersten Zahnprofils der ersten Vielzahl von Zähnen des ersten Getriebes (10) und dem zweiten im allgemeinen hohlen Abschnitt (12b) des zweiten Zahnprofils der zweiten Vielzahl von Zähnen des Gegenrads (12).

8. Getriebesystem gemäß Anspruch 7, **dadurch gekennzeichnet, daß** das erste Getriebe (10) und das Gegenrad (12) schraubenförmig sind.

9. Getriebesystem gemäß Anspruch 8, **dadurch gekennzeichnet, daß** ein Verhältnis der Winkelgeschwindigkeit des ersten Getriebes (10) zu der Winkelgeschwindigkeit des Gegenrads (12) ein Festwert ist.

10. Getriebesystem gemäß Anspruch 1, **dadurch gekennzeichnet, daß** das erste Getriebe (10) ein einfaches Gestellprofil aufweist, das einen Übergangsbereich aufweist, der dem ersten Übergangsbereich (10a) des ersten Zahnprofils der ersten Vielzahl von Zähnen des ersten Getriebes (10) entspricht, welcher eine stetige Kurve ist.

11. Getriebesystem gemäß Anspruch 10, **dadurch gekennzeichnet, daß** die stetige Kurve des Übergangsbereichs des einfachen Zahnstangenprofils geradlinig ist.

12. Das einfache Zahnstangenprofil gemäß Anspruch 10, **dadurch gekennzeichnet, daß** die stetige Kurve des Übergangsbereichs des einfachen Zahnstangenprofils eine kubische Kurve ist.

13. Getriebesystem gemäß Anspruch 1, **dadurch gekennzeichnet, daß** kein Kontakt beim Eingreifen der ersten Vielzahl von Zähnen des ersten Getriebes (10) in die zweite Vielzahl von Zähnen des Gegenrads (12) entlang dem ersten Übergangsbereich (10a) und dem zweiten Übergangsbereich (12a) erfolgt.

14. Getriebesystem gemäß Anspruch 7, **dadurch gekennzeichnet, daß** kein Kontakt beim Eingreifen der ersten Vielzahl von Zähnen des ersten Getriebes (10) in die zweite Vielzahl von Zähnen des Gegenrads (12) entlang dem ersten Übergangsbereich (10a) und dem zweiten Übergangsbereich (12a) erfolgt.

Revendications

1. Un système d'engrenage comprenant :

une première roue (10) ayant une première pluralité de dents ayant chacune un premier profil de dent comportant une première zone de transition (10a) disposée entre une première portion concave (10b) se trouvant dans le creux de dent de ladite première roue (10) et une première portion convexe (10c) se trouvant dans la saillie de dent de ladite première roue (10) ; et

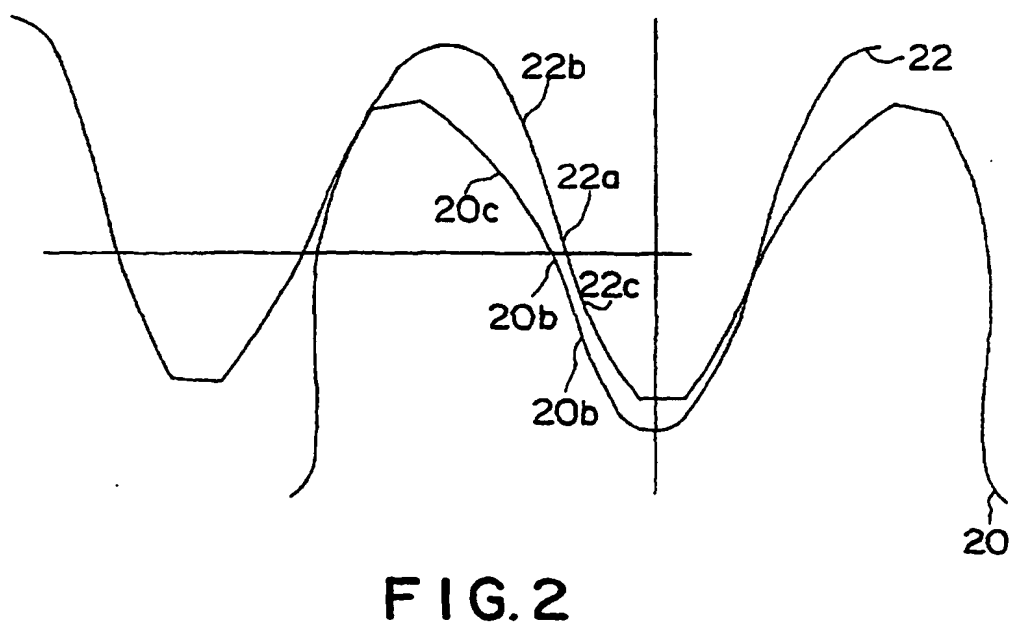
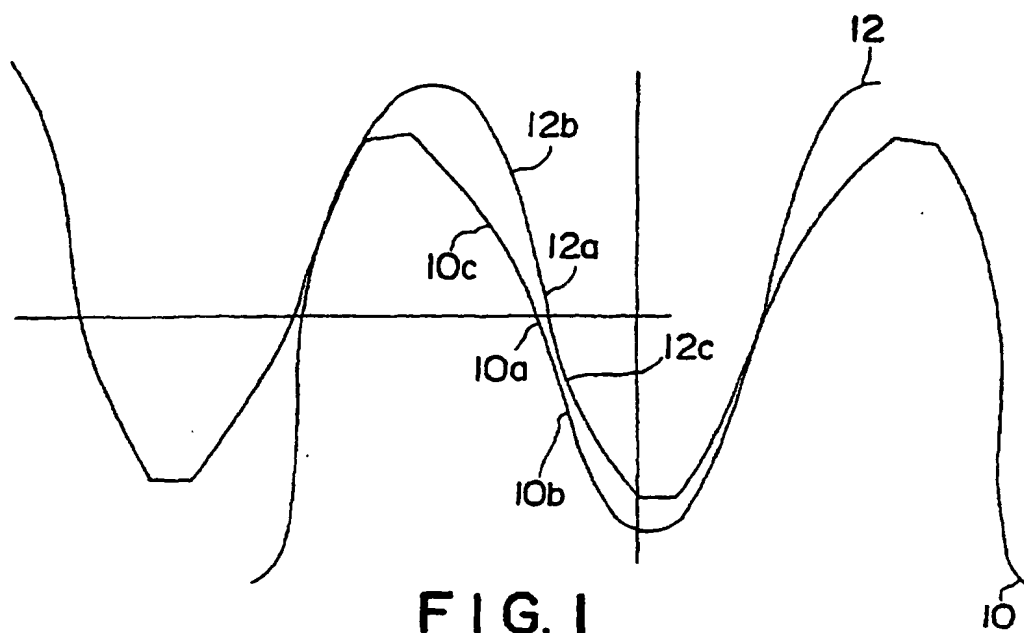
une roue conjuguée (12) ayant une deuxième pluralité de dents ayant chacune un deuxième profil de dent comportant une deuxième zone de transition (12a) disposée entre une deuxième portion concave (12b) se trouvant dans le creux de dent de ladite roue conjuguée (12) et une deuxième portion convexe (12c) se trouvant dans la saillie de dent de ladite roue conjuguée (12),

caractérisé en ce que :

(1) ledit premier profil de dent de ladite première pluralité de dents de ladite première roue (10) est conjugué audit deuxième profil de dent de ladite deuxième pluralité de dents de ladite roue conjuguée (12), sauf dans lesdites première et deuxième zones de transition (10a, 12a) où ladite deuxième portion concave (12b) est conjuguée à ladite première portion convexe (10c) et ladite deuxième portion convexe (12c) est conjuguée à ladite première portion concave (10b), et

(2) la courbure relative dudit premier profil de dent de ladite première pluralité de dents de ladite première roue (10) et dudit deuxième profil de dent de ladite deuxième pluralité de dents de ladite roue conjuguée (12) est une constante qui est inférieure à la courbure relative de roues à développante à angle de pression de 20° au niveau de leur point primitif.

2. Le système d'engrenage de la revendication 1, **caractérisé en ce qu'une** fonction mathématique prédéterminée que paramètrent un rayon de courbure associé audit premier profil de dent de ladite première pluralité de dents de ladite première roue (10) et un rayon de courbure associé audit deuxième profil de dent de ladite deuxième pluralité de dents de ladite roue conjuguée (12) est une constante.
3. Le système d'engrenage de la revendication 1, **caractérisé en ce que** la contrainte de contact maximale lors de la mise en prise de ladite première pluralité de dents de ladite première roue (10) avec ladite deuxième pluralité de dents de ladite roue conjuguée (12) est une constante.
4. Le système d'engrenage de la revendication 1, **caractérisé en ce que** l'axe de rotation de ladite première roue (10) est parallèle à l'axe de rotation de ladite roue conjuguée (12).
5. Le système d'engrenage de la revendication 1, **caractérisé en ce que** l'axe de rotation de ladite première roue (10) forme un angle avec l'axe de rotation de ladite roue conjuguée (12).
6. Le système d'engrenage de la revendication 1, **caractérisé en ce que** ledit premier profil de dent de ladite première pluralité de dents de ladite première roue (10) est un premier profil de coupe transversale, et **en ce que** ledit deuxième profil de dent de ladite deuxième pluralité de dents de ladite roue conjuguée (12) est un deuxième profil de coupe transversale.
7. Le système d'engrenage de la revendication 1, **caractérisé en ce que** le contact est établi, lors de la mise en prise de ladite première pluralité de dents de ladite première roue (10) avec ladite deuxième pluralité de dents de ladite roue conjuguée (12), le long :
 - (a) de ladite première portion concave (10b) dudit premier profil de dent de ladite première pluralité de dents de ladite première roue (10) et de ladite deuxième portion convexe (12c) dudit deuxième profil de dent de ladite deuxième pluralité de dents de ladite roue conjuguée (12), et
 - (b) de ladite première portion convexe (10c) dudit premier profil de dent de ladite première pluralité de dents de ladite première roue (10) et de ladite deuxième portion généralement concave (12b) dudit deuxième profil de dent de ladite deuxième pluralité de dents de ladite roue conjuguée (12).
8. Le système d'engrenage de la revendication 7, **caractérisé en ce que** ladite première roue (10) et ladite roue conjuguée (12) sont hélicoïdales.
9. Le système d'engrenage de la revendication 8, **caractérisé en ce qu'un** rapport entre la vitesse angulaire de ladite première roue (10) et la vitesse angulaire de ladite roue conjuguée (12) est une constante.
10. Le système d'engrenage de la revendication 1, **caractérisé en ce que** ladite première roue (10) a un profil de crémaillère de référence ayant une zone de transition correspondant à ladite première zone de transition (10a) dudit premier profil de dent de ladite première pluralité de dents de ladite première roue (10) qui est une courbe lisse.
11. Le système d'engrenage de la revendication 10, **caractérisé en ce que** ladite courbe lisse de ladite zone de transition dudit profil de crémaillère de référence est une ligne droite.
12. Le profil de crémaillère de référence de la revendication 10, **caractérisé en ce que** ladite courbe lisse de ladite zone de transition dudit profil de crémaillère de référence est une courbe cubique.
13. Le système d'engrenage de la revendication 1, **caractérisé en ce qu'aucun** contact n'est établi, lors la mise en prise de ladite première pluralité de dents de ladite première roue (10) avec ladite deuxième pluralité de dents de ladite roue conjuguée (12), le long de ladite première zone de transition (10a) et de ladite deuxième zone de transition (12a).
14. Le système d'engrenage de la revendication 7, **caractérisé en ce qu'aucun** contact n'est établi, lors de la mise en prise de ladite première pluralité de dents de ladite première roue (10) avec ladite deuxième pluralité de dents de ladite roue conjuguée (12), le long de ladite première zone de transition (10a) et de ladite deuxième zone de transition (12a).



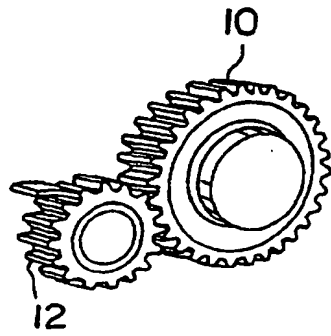


FIG. 3A

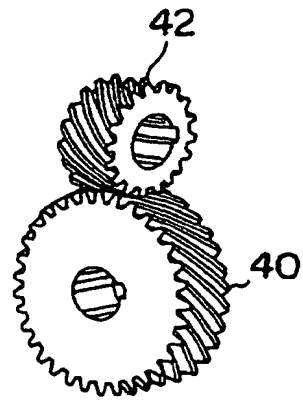


FIG. 3B

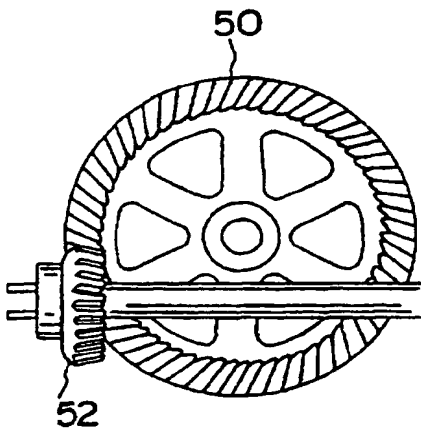


FIG. 3C

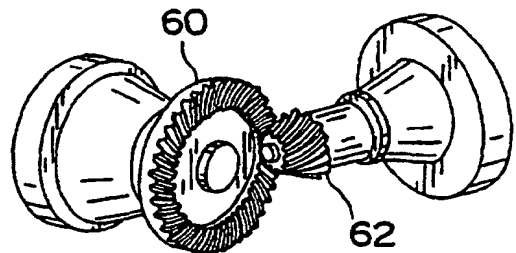


FIG. 3D